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THERMAL ANALYSIS OF A SOLAR AIR HEATER INTEGRATED WITH FINNED ABSORBER PLATE

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ABSTRACT

In the given analysis, the exergetic analysis of finned absorber plate solar air heater (SAH) is done. Here Thermal model for finned based absorber plate SAH is developed. Experimental observations were made in Indian (25° N, 81° E) climatic circumstances. About 9.87% deviation is observed in theoretical & experimental values of exit air temperature of SAH. Highest value 2.17 % exergy efficiency could be accomplished here. The obtained results show the improvement in the solar air heater performance.

KEYWORDS: Solar Collectors, Fin Efficiency, Total Losses & Exergy Destruction

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Original Article

1. INTRODUCTION

The development of a country depends upon its capability to use existing sources of energy. In present day's India's progress towards agriculture, transport, & industrial sectors is remarkable. The maximum pares to India's energy consumption are meet by the fossil fuels. The pace of depletion of resources is extremely high & it will be continued for the long period [1, 2, and 3]. Thus, need for alternative source, as energy becomes equally important. Solar energy is the best gifted clean energy source & emits energy at the pace of $38x10^{22}$kW, of which, just about $18x\ 10^{13}$kW is intercepted on the earth [4]. The simplest & the most proficient way to utilize clean energy by transforming the thermal energy for heating application' through solar collectors. SAH'S, because of their ease, it is cheap & most widely used [5]. The thermal conduct of solar air collector's is relying upon the material, shape, dimensions & layout of collector's. Upgrading in Performance could be accomplished by varying above parameters. To raise the heat transfer in absorber plate & the air, the modification can be done easily.

This includes the use of absorbers - (corrugated absorber or matrix type absorber), with (packed bed or baffles or fins). Thermal analysis is an effective means to obtain accurate and important information to know energy efficiency & losses because of irreversibility in a real state [6]. The first law is mostly used in engineering practice & it depends on the heat balance technique i.e. universally used in the performance investigation. The second law involves the reversibility or irreversibility of process & it is a very fundamental aspect of the exergy & energy investigation's [7]. Exergy data's are more practical & realistic in comparison to the respective energy values; hence exergy analysis provides a more realistic view of processes, sometimes noticeably different in comparison to

the standard energy analysis [8-9].

2. EXPERIMENTAL SETUP

SAH with Single pass is fabricated. Black coated a galvanized iron plate of small thickness is used. Fin's are attached at the back of absorber plate (at perpendicular position & also maintained a minor gap from the base plate) to improve thermal performance in SAH. It performs in the forced convection mode & the fan placed at the inlet of set up. The SAH is insulated using glass wool at the base & sides, to reduce respective heat losses. Here the mass flow rate of 0.036Kg/sec, 0.039Kg/sec & 0.039Kg/sec is maintained by regulator all over the study. Solar intensity & wind velocity is recorded throughout solary meter & anemometer correspondingly.

A temperature at the different point is measured through k-type of the thermocouple. Schematic view of finned absorber plate SAH is given in Figure 1. All relevant parameters & dimensions are listed in table 1.

Number of fins = 14

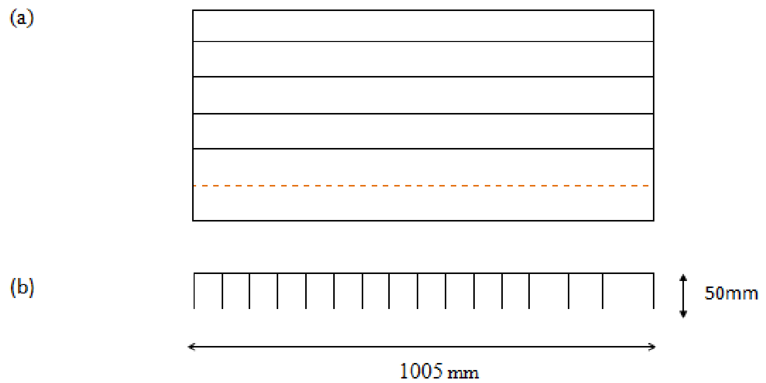

Figure 1: View of Finned SAH

The experiments are performed in the Solar Energy Laboratory of the Mechanical Engineering Department, SIET, SHUATS Allahabad. The experiments were conducted in the month of October in winter climatic condition & observations are recorded between 8 am to 5:00 pm.

3. MATHEMATICAL MODELING

(a). Energy Analysis

The photograph of the finned absorber plate SAH under analysis is given in Figure 1. For the given analysis, energy balance equations are given for solar input, the heat & mass flow rates across the collector, & thermal losses. Heat transfer modes & heat exchanges in the collector are presented in Figure 2. Energy input to the system through the solar radiation impinges on absorber surface. The total losses from the system are due to convection & radiation losses from absorber surface & back plate [10, 11].

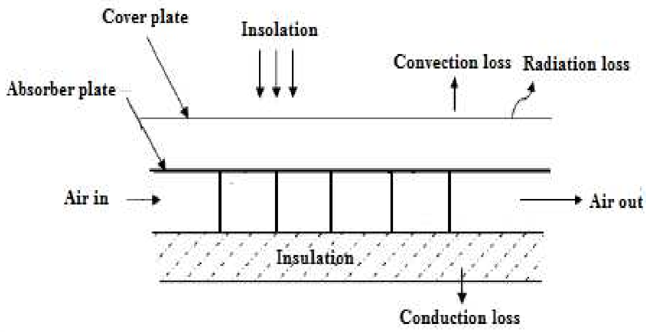

Figure 2: Energy Balance Presentation Absorber Plate

The energy analysis of flat plate SAH is given by sukhatme (1993). The useful heat gain:

$$Q_u = \dot{m} C_p \Delta T \tag{1}$$

Hottel - the Whiller equation for useful heat gain from the solar collector is:

$$Q_u = A_c F_R \varpi \tag{2}$$

Temperature of air at the exit of the collector is obtained as:

$$T_o = T_i + \frac{1}{u_L} \varpi \Upsilon \tag{3}$$

Thermal Efficiency estimated in solar collector:

$$\eta_c = \frac{Q_u}{I_T A_p} \tag{4}$$

Finned Collector: A proposed finned collector shown in Figure 2. Space between fins W_1 & the fins length W_2 can vary widely. Here the fins are attached on the reverse side of absorber plate & there is a little gap between these fins & the bottom plate.

Efficiency factor F' in finned collector: [12].

$$F' = F'_o \left[1 + \frac{1 - F_o'}{\frac{F_o'}{F_P} + \frac{W_1 h_1}{2 W_2 h_2 F_F}} \right] \tag{5}$$

(b) Exergy Analysis

By applying exergy equilibrium in SAH as shown in Figure.1, Exergy balance in flat plate SAH can be expressed as Chamoli (2013):[13]

$$\Psi_{in} - \Psi_{out} - \Psi_{loss} - \Psi_{change} - \Psi_{des} = 0 \tag{6}$$

Considering inlet & exit exergy , the exergy efficiency expressed:

$$\eta_{\Psi} = \frac{\dot{m}}{I_T A_p \eta_p}\left[C_p\left(T_o - T_i - T_a l_n \frac{T_o}{T_1}\right)\right] \quad (7)$$

Exergy Destruction: Exergy destruction caused by irreversibility's in the system. The destroyed exergy rate incorporate following three terms:

One is due to temperature difference between absorber plate surface & sun:

$$\Psi_{des,abs} = \eta_o I_T A_p T_a \left(\frac{1}{T_p} - \frac{1}{T_s}\right) \quad (8)$$

Second term caused by duct's pressure drop:

$$\Psi_{des,\Delta p} = \frac{\dot{m}}{\rho} \times \frac{\Delta_p T_a l_n \left(\frac{T_o}{T_a}\right)}{(T_o - T_{in})} \quad (9)$$

Third term is due to temperature difference between absorber plate surface & agent fluid:

$$\Psi_{des,conv} = \dot{m} C_p T_a \left(l_n\left(\frac{T_o}{T_{in}}\right) - \frac{(T_o - T_{in})}{T_p}\right) \quad (10)$$

Thus exergy destruction rate:

$$\Psi_{des} = \frac{\Psi_{des,abs} + \Psi_{des,\Delta p} + \Psi_{des,conv}}{\Psi_{in,r}} \quad (11)$$

Since the duct pressure drop is neglected in the present case so that the rate of exergy destruction is:

$$\Psi_{des} = \frac{\Psi_{des,abs} + \Psi_{des,conv}}{\Psi_{in,r}} \quad (12)$$

Exergy In: According to Petela's, exact exergy in come by solar radiation on the typical collector of surface area A_p becomes:

$$\Psi_{in,r} = I_T A_p \eta_p \quad (13)$$

(c) Parameter Range

Table 1: Input Parameters & Their Values used in the Study

S. No	Input Parameter	Range
1	Glazing	Single glass
2	Fluid	Air
3	Number of Finns	5
4	Length of plate	L=1.21m
5	Width of plate	W=0.92m
6	Depth of channel	s=0.05m
7	Sun Temperature	5760K
8	Transmttance-absoptance	0.84

Table 1: Contd.,		
9	plate thickness δ	0.002m
10	Absorber plate emittance, εp	0.92
11	ǀlass cover emmitance, εg	0.88
12	tefan-Boltzman constant	5.67x10-8 W/m2 K4
13	pace between fins W1	0.0657m
14	Length of fin W2	.05m

Heat Transfer Coefficients: Rate of heat transfer for convective & radiative is analyzed by the following ways: [14] for calculating the radiative heat transfer:

$$h_{r,p-g} = \frac{\sigma\left(T_p + T_g\right)\left(T_p{}^2 + T_g{}^2\right)}{\left(\frac{1}{\varepsilon_p}\right) + \left(\frac{1}{\varepsilon_g}\right) - 1} \tag{14}$$

Sherwin suggested in laminar flow, can be modeled for flat plate & proposed relations to heat transfer coefficient:

For $R_{eL} \le 5\times10^5$ *and* $\left(P_r \ge 0.5\right)$, $N_u = 0.664\, P_r^{\frac{1}{3}} R_{eL}^{\frac{1}{2}}$

For turbulent flow, Niles et al. proposed the following relationship:

For $5\times10^5 < R_{eL} < 10^8$ and $0.6 < P_r < 60$, $N_u = 0.333\, R_e^{0.8} P_r^{0.33}$

Finally convective heat loss coefficient can be calculated by using the above conditions.

$$h_c = \left(\tfrac{k}{L}\right)\times0.664\times\left(P_r\right)^{\frac{1}{3}}\times\left(R_e\right)^{\frac{1}{2}} \tag{15}$$

Loss Coefficients: Top loss coefficient in case of single glass cover system [15, 16] :

$$U_t = \left(\frac{1}{h_{cp-c} + h_{rp-c}} + \frac{1}{h_w + h_{rc-a}}\right)^{-1} \tag{16}$$

Total loss: $U_L = U_t + U_b + U_e$ (17)

4. RESULT AND DISCUSSIONS

Energy balance & rate equations discussed earlier to find the exit air temperature, collector efficiency and heat delivered for given set of input values, for given time span. Intermediate values would include estimation of heat transfer coefficients & heat flux between air heater components. For subsequent times, initial temperature specified equal to temperatures measured in that of previous sections. The iterative process continued until the end time is reached. As the initial condition, absorber plate is black & air temp. is assume as ambient temperature.

Experiments were conducted in the winter seasons of Indian climatic conditions and in the premises of SHUATS Allahabad. Observations were taken during the whole month and best value of a particular day is used in the study. Figure.3 shows the variation of solar intensity with respect to time of the day. Solar intensities are almost close for different mass flow rates and vary in the same way.

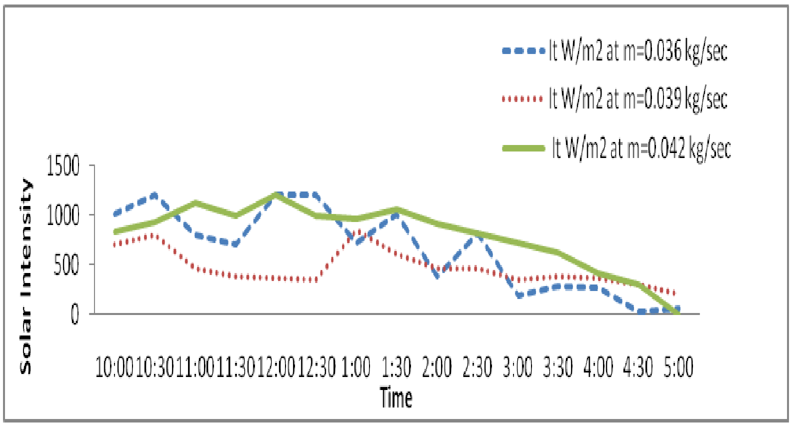

Figure 3: Variation of Solar Intensity at Different Mass Flow rate with Respect to Time of the Day

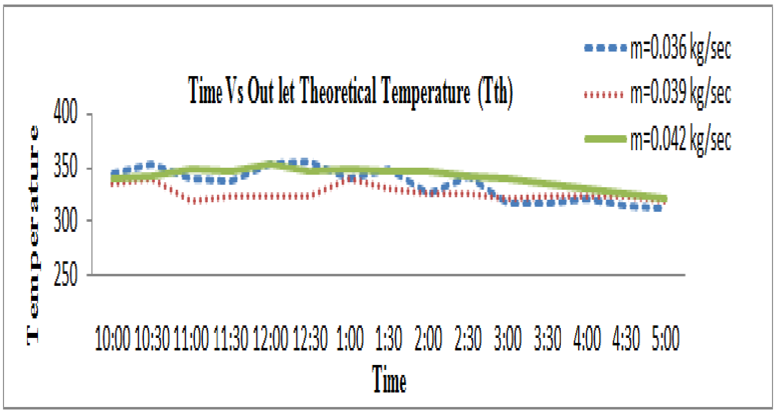

Figure 4: Variation of Outlet Theoretical Temperatures at Different Mass Flow rate with Respect to Time of the Day

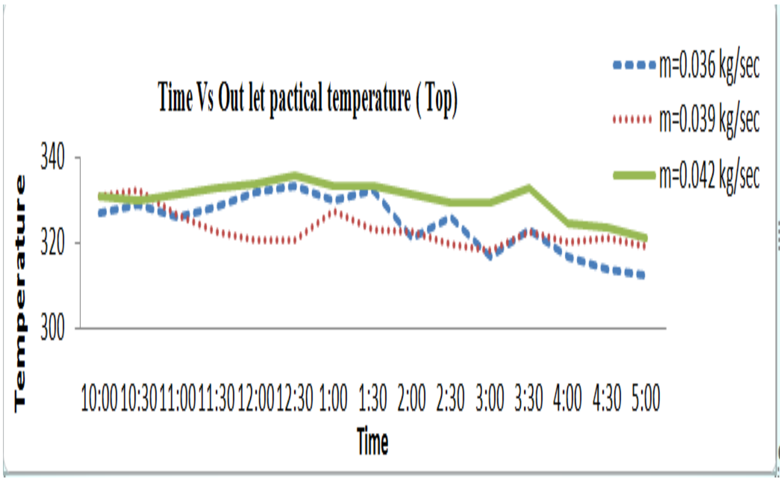

Figure 5: Variation of Outlet Practical Temperatures at Different Mass Flow rate with Respect to Time of the Day

Figure 4 and Figure 5 show the variation of exit temperatures at mass flow rates of 0.036 kg/s, 0.039 Kg/sec and 0.042 kg/s theoretically and practically. Exit temperatures also vary in the same way as that of the solar intensity. Theoretical values differ by their corresponding experimental values within a limit of 10%.

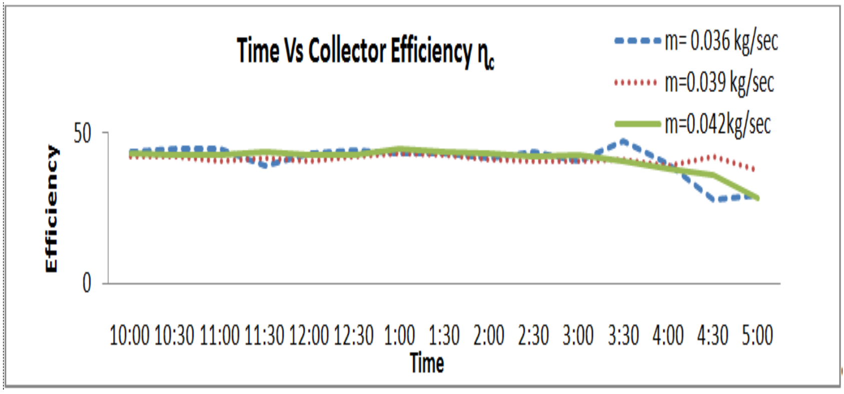

Figure 6: Variation of Collector Efficiencies W. R. T. Time of the Day at the Different Mass Flow Rate

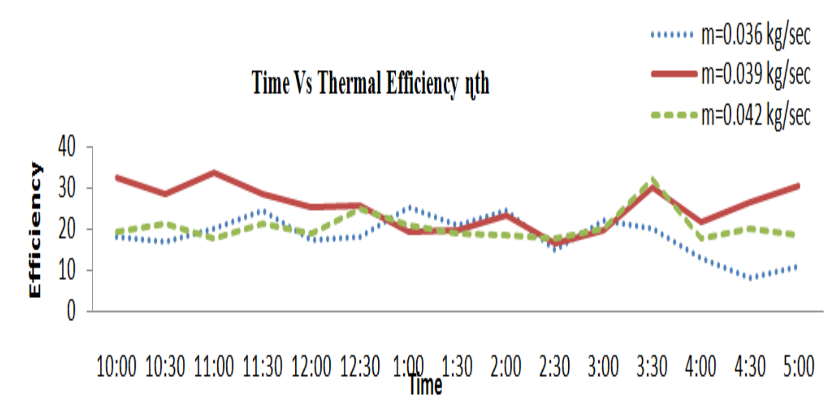

Figure 7: Variation of Energy Efficiency W. R. T. Time of the Day at Different Mass Flow Rate

Figure 6 shows variations of collector efficiency at the three mass flow rates with respect to time of the day (hr). As it is expected, a very small variation is obtained at the overall average collector mass flow rates. The average collector efficiency for all three mass flow rates found to be almost same as per the trend. Figure 7 show the variation of thermal efficiency at the different mass flow rates of 0.036 kg/s. 0.039kg/sec and 0.042 kg/s with respect to time of the day. When the mass flow rate is increased from 0.036 kg/s to 0.042 kg/s increase in energy efficiency is obtained as 10.84%. Higher energy efficiency is obtained at the higher mass flow rate. It is due to the fact that at the higher mass flow rate convective heat transfer rate to the air will be higher. Energy utilization will be more and the loss to the environment will be small.

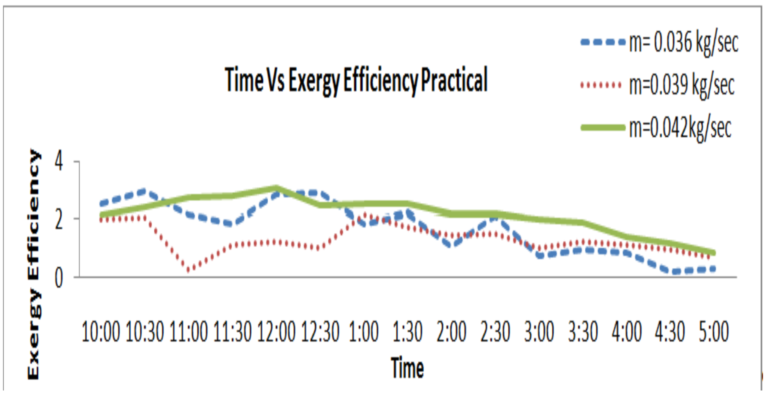

Figure 8: Variation of Exergy Efficiency Practical with Respect to Time of the Day (hr) at the Different Mass Flow Rate

It is clear from the Figure 8, where practical exergy efficiency at higher mass flow rate is more. Exergy loss is reduced at the higher mass flow rate. When the mass flow rate is increased from 0.036 kg/s to 0.042 kg/s exergy efficiency increases by 51%. The average practical exergy efficiency could achieve 1.724 %, 1.307 & 2.17 % at mass flow rate of 0.036, 0.039 & 0.042 Kg/sec. respectively.

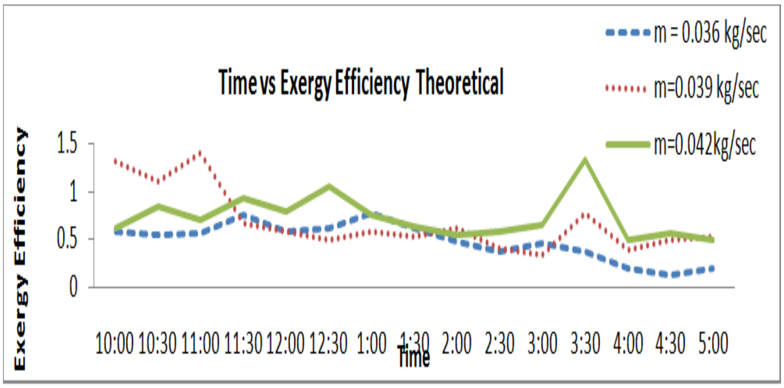

Figure 9: Variation of Exergy Efficiency Theoretical with Respect to Time of the Day (hr) at the Different Mass Flow Rate

It is clear from the Figure 9 where theoretical exergy efficiency at higher mass flow rate is more. Exergy loss is reduced at the higher mass flow rate. When the mass flow rate is increased from 0.036 kg/s to 0.042 kg/s exergy efficiency increases by 46%. The average practical exergy efficiency could achieve 0.4785 %, 0.681 % & 0.7242 % at mass flow rate of 0.036 Kg/sec, 0.039 Kg/sec & 0.042 Kg/sec. respectively.

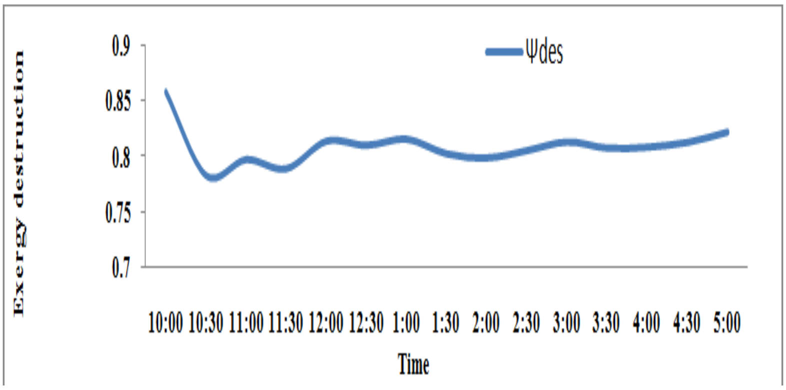

Figure 10: Variation in Exergy Destruction, w. r. t. Time

Figure 10 predicts the trend of the exergy destruction with respect to the time of the day. The maximum average instantaneous destruction found to be 0.86 and the overall average destruction is found to be 0.81.

5. CONCLUSIONS

This work presents the details about the Mathematical model of a SAH using heat transfer expressions to collector components & empirical relations for estimating different Heat transfer coefficients. It predicts thermal performance in SAH over a wide range of operating condition namely solar intensity & inlet air temperature. Middling collector efficiency of 41.11 % & maximum instantaneous thermal efficiency around 33.84 % is obtained. Exergy analysis shows maximum exergy loss & exergy destruction in the system. Average exergy efficiency of 0. 69% is obtained in the study and the average destruction is about 0.81.

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APPENDICES

Nomenclature:
A Area (m^2)
C_p Specific heat capacity of the fluid (K j /Kg k)
F' Finned collector efficiency factor
F_R Heat removal factor
I_T Incident Solar radiation (W/m^2)
U_L Over all heat loss coefficient (W/m^2-K)
T Temperature (K)
Q Heat transfer rate (Watt)
D Hydraulic diameter (m)
U Loss Coefficients(W/m^2-K)
T Temperature ^oK
E, H, Value Contant (W/m^2-K)
h_1, h_2, h_3, hc, convective Heat transfer coefficient (W/m^2-K)
S Absorbed solar radiation (W/m^2)
R_e Reynolds's No.
N_u Nusselt No
P_r Prandtl No
L Collector length (m)
W Collector width (m)
s Depth of air channel (m)
$\dot{m}$ Mass flow rate (Kg/sec)
h_w Convective heat transfer due to wind.
F Fin efficiency
Greek symbols :
a absorptance
η Efficiency
τ Transmittance
(τα) Transmittance-absoptance
σ Stefan-Boltzman constant
ε emmitance
μ Dynamic viscosity (Kg/m-s)
k Thermal conductivity (W/m-k)
Ψ Exergy
Subscripts:
a ambient
o outlet . optical
i inlet
exp exponential
p plate , petla
c collector
u useful
f fin
t_h Thermal
s Sun, Depth of channel
g Glass
c Convective
r Radiative
t Total, top
b Back
e End
r Radiation
abs Absorber
conv Convective

$\Delta T = (T_i - T_a)$	$F'_o = \frac{E}{\left[H+U_b(h_2+h_rU_t)+U_t(h_r+h_3)\right]}$	$F_F = \frac{\tanh(mW_2)}{mW_2}$	$U_b = \frac{K}{l}$
$\varpi = \left[S - u_L(T_i - T_a)\right]$	$E = H + h_2U_b$	$m = \sqrt{\frac{h_2}{k\delta}}$	$U_e = \frac{(UA)_{edge}}{A_c}$
$\Upsilon = \left[1-\exp\left\{-\frac{A_c u_L F'}{\dot{m} C_p}\right\}\right]$	$H = \left(h_2h_3 + h_2h_r + h_3h_r\right)$	$\eta_p = \frac{4T_a}{3T_s} + \frac{1}{3}\left(\frac{T_a}{T_s}\right)^4$	$A_c = s \times W$
$F_R = \frac{\dot{m} C_p}{A_c u_L}\Upsilon$	$F_p = \frac{h_1}{h_1+U_t}$	$h_w = 5.7 + 3.8 \times V$	$D = 4\left[\frac{W\times s}{2W}\right] = 2s$

Computer Model Flow Chart